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## ABSTRACT

This paper presents Nissan's new four-valve-per-cylinder direct injection (DI) diesel engine series consisting of a 2-liter class and 3-liter class. These engine series provide substantially improved power output along with lower noise and vibration levels, which have been traditional drawbacks of DI diesel engines. Nissan developed this engine series in response to the heightened need in recent years for passenger-car DI diesel engines with superior thermal efficiency, a characteristic advantageous for reducing $CO_2$ emissions.

## DESIGN CONCEPT

These engine series are the first high-speed DI diesel engines for Nissan. At the design conceptualization stage of the new DI diesel engine series, it was projected that dramatic improvements would be needed in automotive engine technology to meet the environment-related changes forecasted for the automotive industry in the year 2000. Those changes are typified by the enforcement of severe exhaust emission standards in Japan and Europe. On the other hand, market demand has risen in recent years for higher power output and lower noise levels. Against this backdrop, the design concept of the new DI engine series was formulated as follows:

1. Higher Power Output: To achieve top-level power output in the same class.
2. Lower noise level: To combine several measures to achieve quietness on a par with gasoline-engine vehicles of the same class.
3. High fuel economy: To make a thorough effort in weight reduction for moving parts and a new combustion concept would achieve outstanding fuel economy.
4. Exhaust emission: To meet emission levels such as the long-term emission regulation in Japan and CEDIII emission regulation in Europe.
5. Lightweight and compact: To adopt lightweight design for moving parts and use analytical design methods to make this engine series lighter. A smaller engine would allow installation into a front engine-front wheel drive car.

## BASIC DIMENSIONS

These engine series are water-cooled, in-line 4, chain-driven DOHC, 16-valve engines. The basic engine dimensions are given in Table 1.

Table 1. Basic Engine Spec.

| | 2-liter class | 3-liter class |
|---|---|---|
| Type | Water-cooled, Diesel | ← |
| Combustion chamber | Direct-Injection | ← |
| Cylinder arrangement | In-line 4 | ← |
| Displacement $cm^3$ | 2488 | 2953 |
| Bore x Stroke mm | 89×100 | 96×102 |
| Compression ratio | 18.0 : 1 | ← |
| Valve train | 4-Valve DOHC | ← |
| Turbo charger | Variable-nozzle, with Inter-cooler | ← |
| Injection pump | BOSCH VP44 | ← |
| Injection nozzle | 2-spring | ← |
| Emission control system | Electronically-controlled EGR system | ← |
| | Oxidation Catalyst | ← |
| Max. Power (Target) | 110kw/4000rpm | 125kw/3600rpm |
| Max. Torque (Target) | 309Nm/2000rpm | 351Nm/2000rpm |

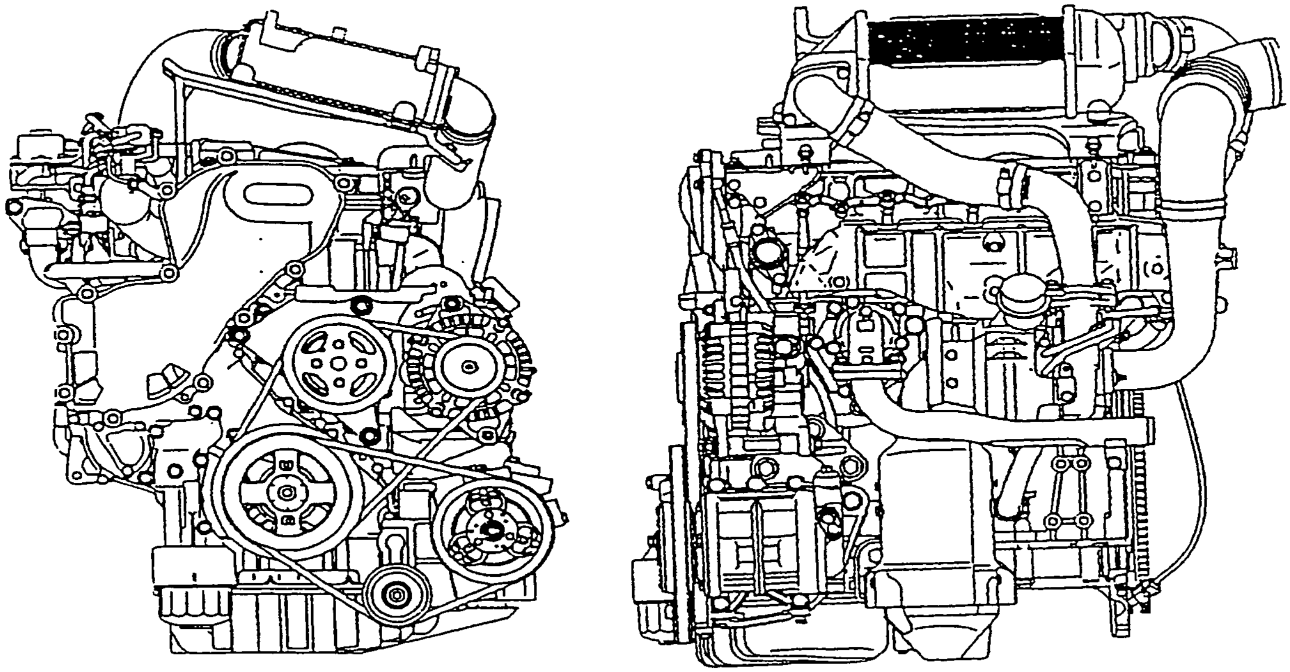

Figure 1. Outline Drawings of the 2-liter Class

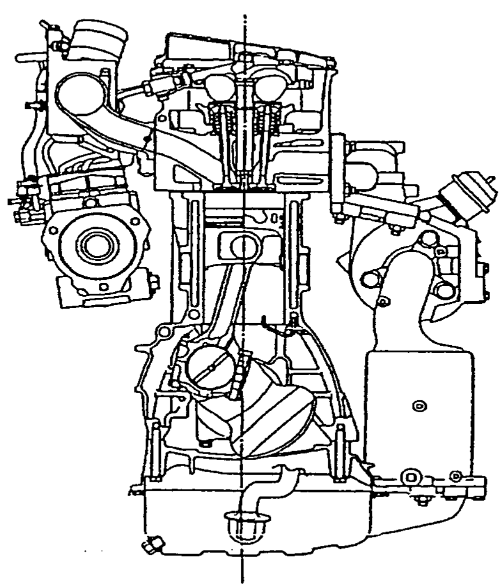

Figure 2. Cross Sectional View of the 2-liter Class

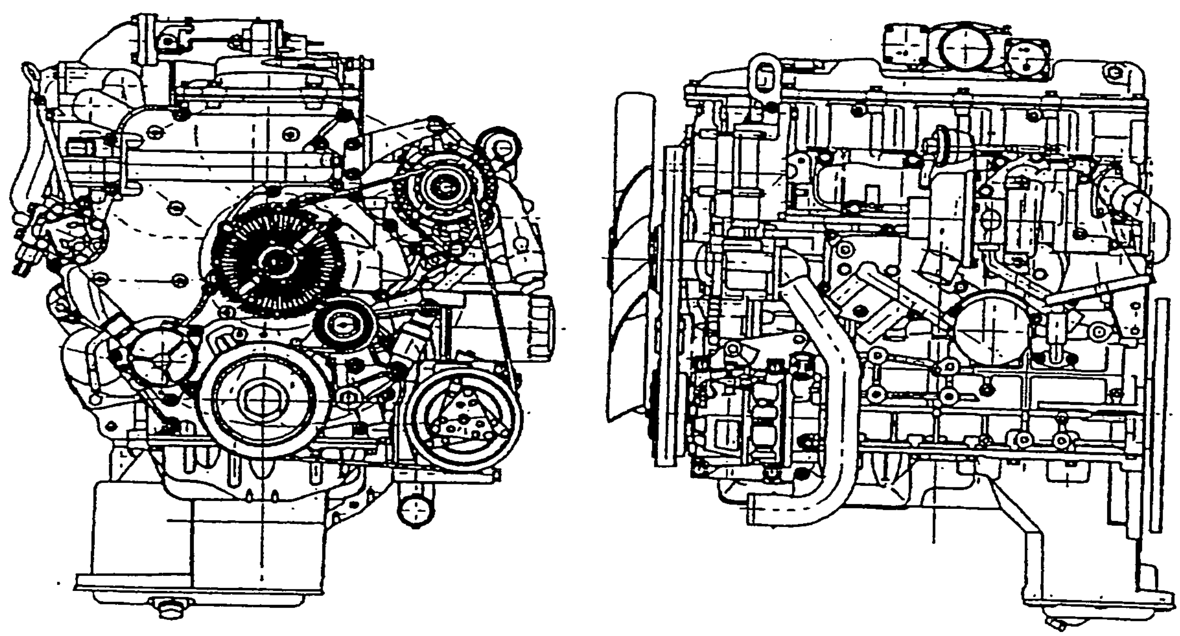

Figure 3. Outline Drawings of the 3-liter Class

Figures 1, 2 show the external shape and the cross-sectional view of the 2-liter class.

Figure 3 shows the external shape of the 3-liter class.

## BASIC STRUCTURAL DESIGN

COMPACT SIZE – The 2-liter class is mounted transversely in a front-wheel-drive layout. Reduction of overall engine length and width make it possible to mount it in an ordinary small-size car. Finally, due to several measures, even with a 2.5-liter maximum displacement we achieved the same package size as the current 2-liter engine.

Reduction of Overall Engine Length – By adopting a long-stroke design (S/B = 1.12), the cylinder block length was reduced. A chain-driven power steering pump system reduced overall engine length.

Reduction of Overall Engine Width and Height – The auxiliary units were placed closer to the engine proper, and the power steering pump is driven by the timing chain. That arrangement reduces the overall width of the engine. The adoption of mechanical direct-acting bucket tappets achieved a compact valve train assembly. In addition a two-stage timing chain system makes it possible to reduce the cam sprocket diameter. As a result of these measures, the overall height has been reduced.

The 3-liter class is a replacement of the OHV type engine. We tried to minimize the size increase caused by the DOHC layout.

Reduction of Overall Engine Length – The auxiliary units are driven by the serpentine layout poly-V rubber belt. That arrangement reduces overall engine length.

Reduction of Overall Engine Height – As with the 2-liter class, direct-acting bucket tappets and a two-stage cam drive system reduce the engine height.

WEIGHT REDUCTIONS – Lighter moving parts, and weight reduction of main structure parts by using analytical methods, makes these engine series lighter.

## MAJOR STRUCTURAL COMPONENTS

CYLINDER BLOCK – The cylinder block is cast iron. The 2-liter class block has a deep-skirt design, and the overall rigidity of the powertrain has been increased by adopting a cast aluminum oil-pan and by coupling the transmission directly to the engine without an intervening rear plate. The 3-liter class block has a half-skirt design with a ladder frame.

CYLINDER HEAD – The cast aluminum cylinder head has four valves per cylinder and an injection nozzle located in the center of the cylinder. This intake port layout and shape achieved a high swirl ratio and high flow coefficient.

PISTONS – Cast aluminum pistons of the thermal flow type are used. With the lightweight design, such as shorter piston pin and smaller skirt, the weight of the piston has been reduced dramatically. Figure 4 shows the piston model of the 2 liter class. Figure 5 shows the weight comparison.

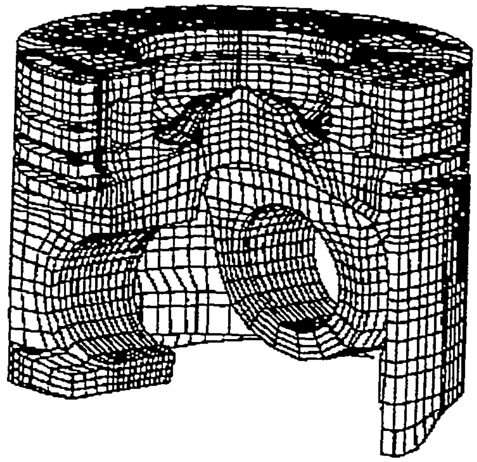

Figure 4. Piston Model of the 2-liter class

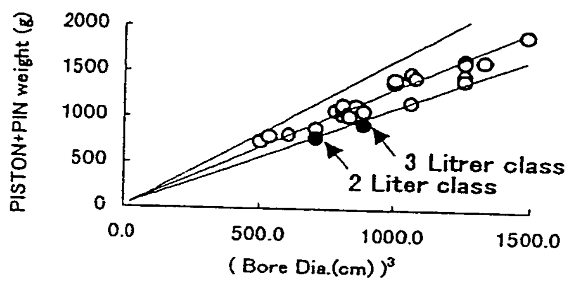


Figure 5. Piston Weight Comparison (DI Diesel Engine)

VALVE TRAIN – The adoption of mechanical direct-acting bucket tappets achieved an exceptionally compact valve train assembly. Hydraulic lash adjusters were eliminated as a result of extensive wear analyses and improvements made to the cam followers. The reduced weight of the valve train made it possible to lower the spring load, which has reduced friction substantially.

CAMSHAFT DRIVE – A two-stage cam drive system was adopted. The 2-liter class uses a two-stage timing chain drive system. The first timing chain system drives the injection pump, and the second timing chain system drives the camshaft. The 3-liter class uses a timing gear drive system on the first stage and a timing chain drive system on the second stage. Both systems make the cylinder head compact.

INTAKE MANIFOLD – The intake manifold is made of cast aluminum and has a thinner wall for weight reduction. The swirl control valves are mounted to achieve sufficient air flow characteristics.

TURBO CHARGER – A variable-nozzle turbo charger is used to improve both low-speed torque and maximum power. The variable nozzle is controlled by an electronically controlled feedback system.

## ENGINE CONTROL

FUEL SYSTEM – An electronically controlled high-pressure distributor type fuel injection pump (VP44) is used. This pump pressurizes fuel up to 1500 bar (147 Mpa), and fuel quantity is controlled by an electromagnetic valve. A two-stage valve opening pressure injection nozzle is used, vertically mounted at the center of the cylinder.

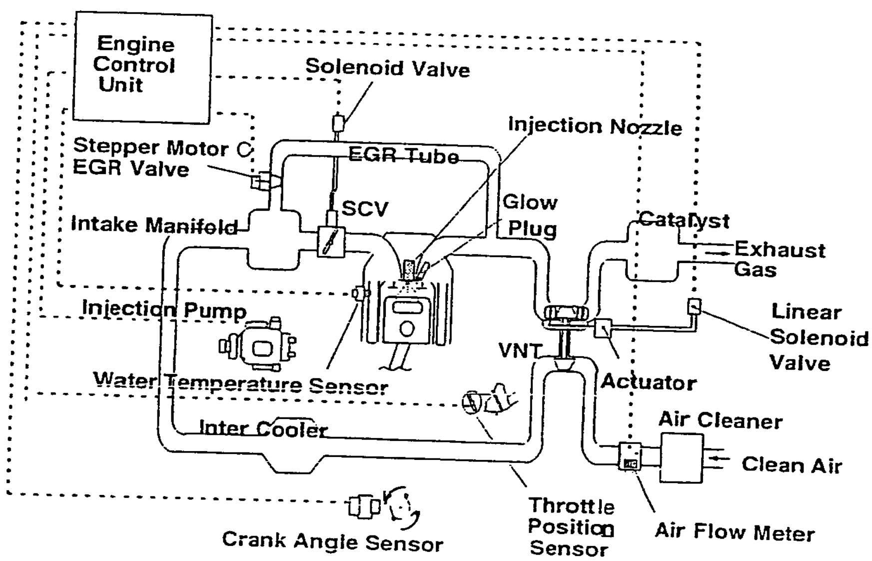


Figure 6. Engine Control System

EMISSION CONTROL – An electronically controlled EGR valve is used to control the heavy EGR rate. This EGR valve is driven by a stepper-motor, and has better response than a vacuum-electronically controlled system. The EGR system is controlled by a feed back system. An oxidation catalyst is also equipped. Figure 6 shows the engine control system.

## PERFORMANCE

POWER OUTPUT – Emphasis was placed on achieving ample power output and easy drivability over a practical driving range while at the same time assuring pleasant response and powerful acceleration over a wide range. To provide such performance, these engine series have been designed to generate high torque even at low speeds and provide a flat torque characteristic without any fall-off in the high-speed range.

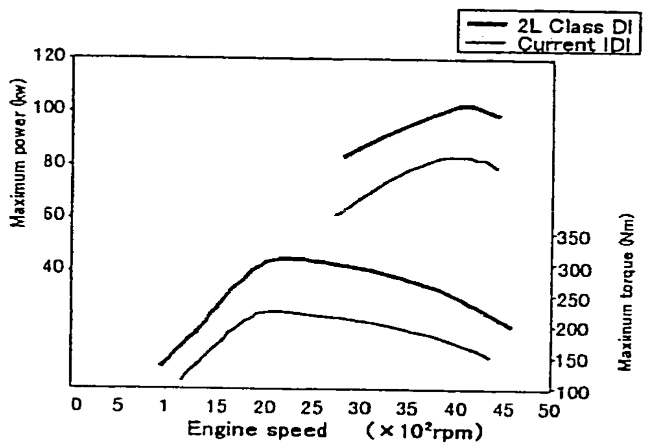


Figure 7. 2-liter Class Power Output Compared with Current IDI Engine

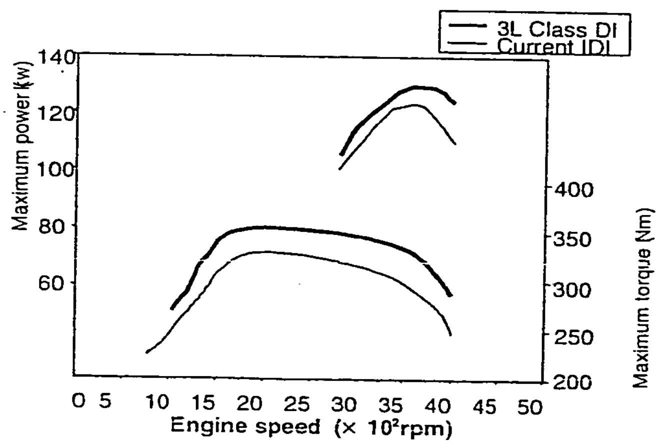


Figure 8. 3-liter Class Power Output Compared with Current IDI Engine

FUEL ECONOMY – Among the various performance objectives set for the new engine, top priority was placed on improving fuel economy. Thorough measures were taken to reduce fuel consumption. As a result, when mounted with one of these engines in a vehicle, Japanese 10-15-mode fuel economy was improved 40% compared with the current IDI diesel engine. A new combustion concept has been adopted to improve fuel economy. Lightweight moving parts and mechanical direct-acting bucket tappets reduce mechanical friction. Torque improvement made it possible to use a higher gear ratio, which improves fuel consumption also. Figure 9 shows fuel economy improvement.

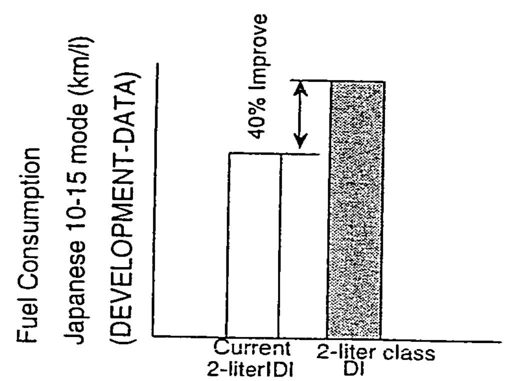


Figure 9. Fuel Economy Improvement (Vehicle Total)

EMISSION PERFORMANCE – To reduce NOx and PM emissions from the engine, a new combustion concept has been adopted. An oxidation catalyst is also equipped to reduce PM and HC. As a result, this engine meets Japanese long-term exhaust emission regulations.

## NOISE AND VIBRATION

REDUCTION OF COMBUSTION NOISE – To reduce combustion noise a new combustion concept has been adopted. A two-stage valve opening pressure injection nozzle was used. This nozzle reduces injected fuel quantity at the early stage of injection, so the rate of pressure increase in the cylinder is lower, which reduces the combustion noise.

REDUCTION OF EXCITATION FORCE – Most of the engine excitation force is produced by the movement of moving parts such as the piston. Consequently emphasis was focused on reducing the mass of moving parts so as to decrease the force generated by their movement. The 3-liter class incorporated a balance shaft.

IMPROVEMENT OF STRUCTURE ATTENUATION – A structural analysis method was utilized to achieve high rigidity of structural parts, such as cylinder block, cylinder head, etc. The 3-liter class adopts a ladder frame for the cylinder block skirt. Figure 10 shows the effect of the ladder frame.

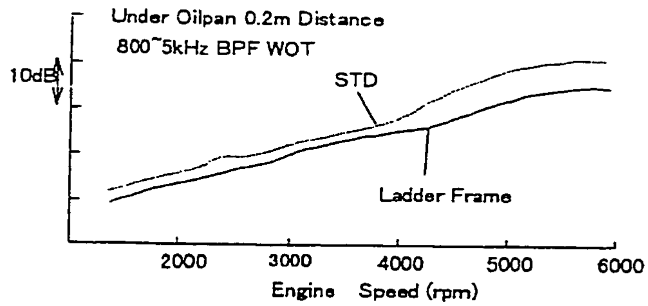

Figure 10. Effect of Ladder Frame

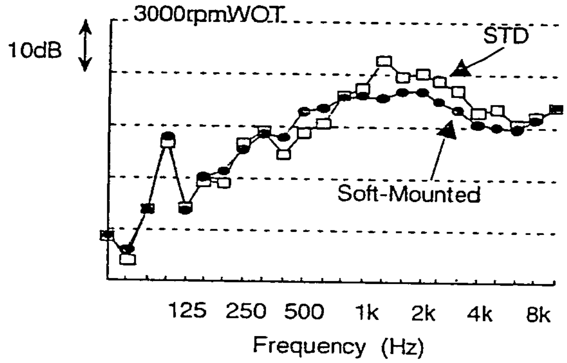

Figure 11. Effect of Soft-Mounted Chaincover

NOISE ISOLATION AND INSULATION – A soft mounted rocker cover and chain cover are used. The 2-liter class has a double-walled oil pan to achieve good damping and isolation performance. A noise insulation cover is also adopted. Figure 11 shows the effect of the soft mounted chain cover.

REDUCTION OF ENGINE VIBRATION TRANSMISSION TO BODY – In addition to suppressing excitation forces produced by engine vibrations, steps were also taken to reduce the transmission of engine vibration to the body. This was accomplished by adopting an active controlled engine mounting (ACM) for the 2-liter class with a transversal mounted layout.

## COMBUSTION IMPROVEMENT

NOx and particulate (PM) in the exhaust of diesel vehicles are a source of atmospheric pollution, and their presence has resulted in stricter exhaust regulations. On the other hand, the superior heat efficiency of diesel engines is clearly desired from the viewpoint of easing global warming. Against this background, we have adopted a new combustion concept in these engine series that simultaneously reduces NOx and PM. This system also achieves a decrease in combustion noise without degrading fuel consumption.

This new combustion concept can be summed up by the phrase "low-temperature, pre-mixed combustion." The basic concept behind this new concept is shown in Figure 12.

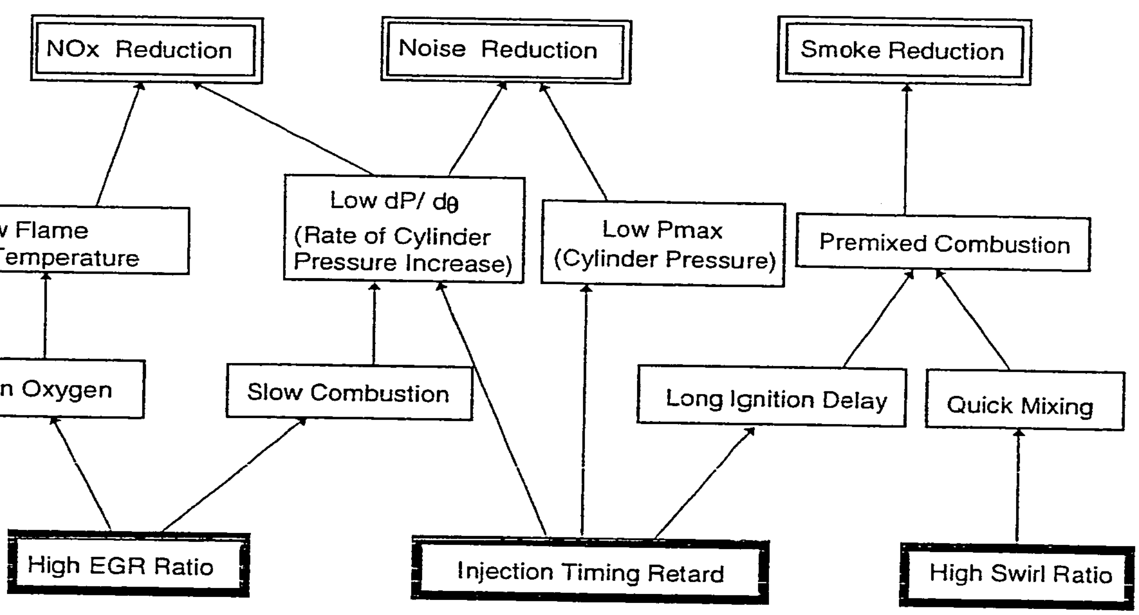

Figure 12. Scheme of New Combustion Concept

The generation of NOx is highly dependent on combustion temperature, and lowering combustion temperature is effective in reducing the amount of NOx. This is approached by lowering the concentration of oxygen in the intake air. One specific way of achieving this is EGR. The new concept reduces PM, which consists of insoluble organic fraction (ISOF), the main components of black smoke and soluble organic fraction (SOF). It appears, however, that the generation of ISF can not be avoided in diffusion combustion, and this is being pursued by pre-mixed combustion. While pre-mixed combustion can be obtained by extending the ignition delay time, there is a concern that early injection will lead to difficulty in ignition timing control and significant increase in HC emissions. At the same time, this new concept is based on the assumption that pre-mixed combustion, while important, is not absolutely essential if deterring from a lean mixture. For these reasons, we have decided to use retarded injection. With this type of injection, however, it is predicted that heat efficiency will degrade and that HC and SOF emissions will increase. We therefore implemented countermeasures to these problems by optimizing both the decrease in cooling loss and the gas flow in the cylinder, by adjusting the shape of the combustion chamber, the value of the swirl ratio, etc.

To verify that this new combustion concept can be used to achieve low-temperature pre-mixing in DI diesel combustion, we observed the combustion with high-speed photography. The results are shown in Figure 13 along with those for standard DI combustion (no EGR and standard injection timing and swirl ratio) for purposes of comparison. Also shown in the figure are heat-release-rate plots for both types of combustion.

Examining heat release rates, it is clear in the case of the new combustion system that the beginning of heat release is later than standard heat release due to significant retardation of injection timing. Another feature of this system is that the heat release rate is relatively low directly after the start of heat release. This is because the rate of pressure increase is kept low. Although the heat release rate is initially low in this new system, it becomes activated in subsequent combustion, and the combustion period is for the most part not inferior to that of standard combustion. Also, with regard to the overall shape of heat-release-rate plots in the figure, that for standard DI takes on a 2-stage format consisting of initial combustion (pre-mixed combustion) and main combustion (diffusion combustion), while that for the new system has a 1-stage format that can be treated overall as pre-mixed combustion.

Focusing our attention on the flame photos, we can see that in the case of the new combustion concept, hardly a bright flame can be observed throughout the combustion cycle. This situation indicates not only that the strength of light emission is weak because of low temperature combustion, but also that the concentration of soot within the flame is low. From the photos we confirm that our original scheme of pre-mixed combustion is working.

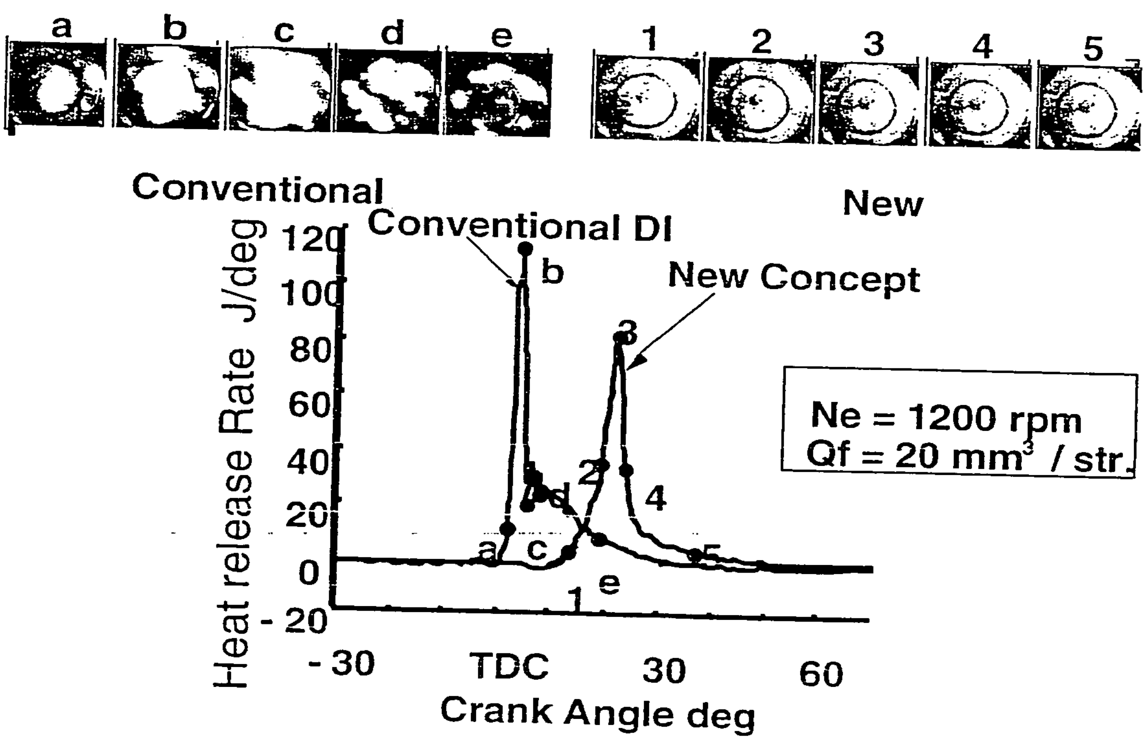


Figure 13. Combustion Flame Photographs and Heat Pelease Rates

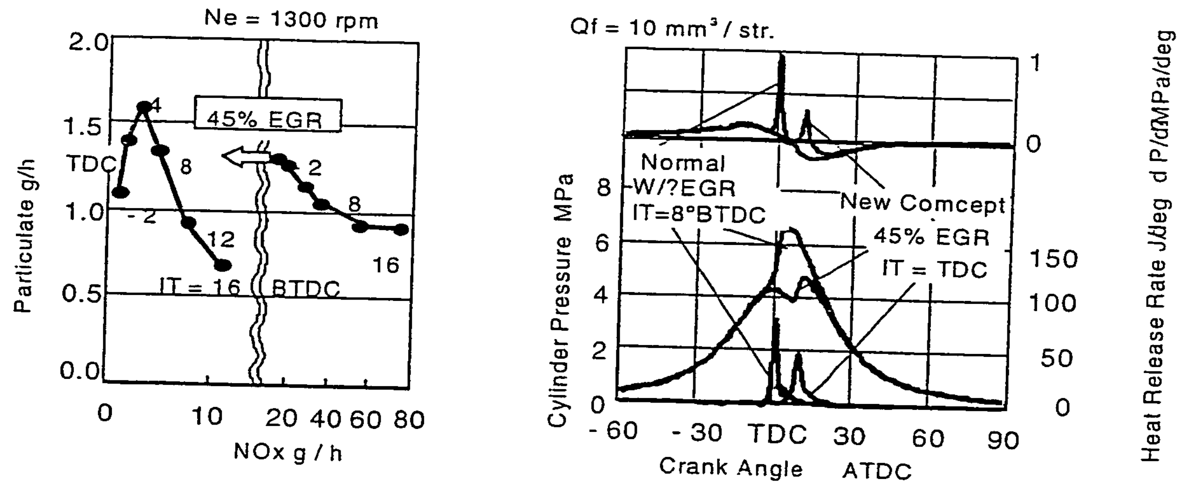


Figure 14. Effects of New Combustion Concept on NOx, PM and Heat Release Rate.

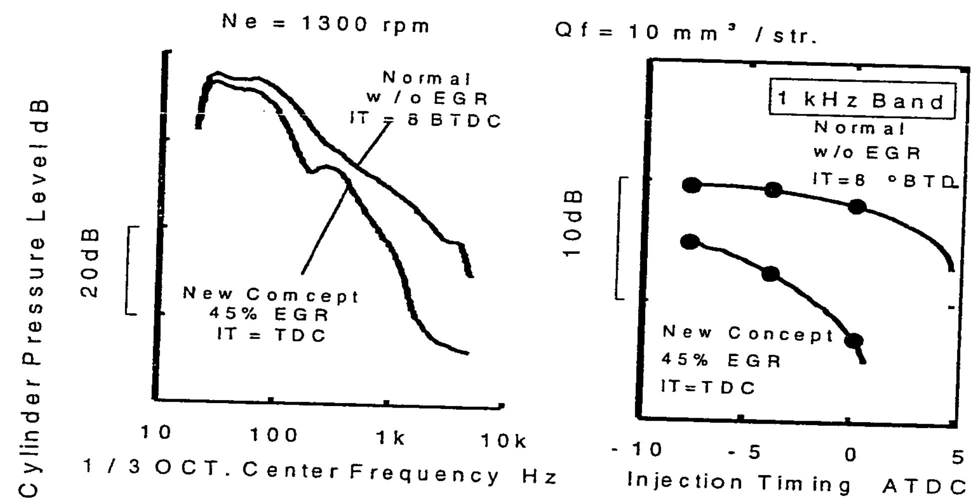


Figure 15. Effects of New Combustion Concept on Cylinder Pressure Level

Figure 14 shows the change in NOx and PM when injection timing is delayed with and without EGR. With no EGR, NOx decreases as injection timing is delayed, but PM increases, which reflects the trade-off relationship mentioned above. With EGR, this trade-off relationship initially appears with delay in injection timing, but as injection timing becomes more retarded PM and NOx simultaneously break into an area where they are both decreasing. As a result, NOx reaches a level under about 1/10 of that at advanced injection timing without degradation in the amount of PM. Examining heat release rates at this time, we see that the initial heat release rate and peak level decreases, and becomes 1-stage combustion. At the same time, we see a significant decrease in both maximum pressure within the cylinder and in the rate of pressure increase. As a consequence, combustion noise is also decreased. Figure 15 shows the effects of the new combustion concept on cylinder pressure level.

## OPTIMIZATION OF INTAKE PORT

To make good use of the new-combustion system described in the previous section, an intake swirl ratio of 10 or greater (measured by an impulse swirl meter) is required. On the other hand, to secure performance at high engine speed a swirl ratio of from 3 to 4 is normally used. Therefore, to make this new combustion concept practical, a variable swirl control system that can control the swirl ratio in the range from 10 to 3 becomes necessary.

We studied intake ports that could achieve such a variable swirl range. As a basic concept for the variable swirl

system, we selected a system consisting of two independent ports, a low-swirl port and a high-swirl port, with a swirl control valve installed at the input of the low-swirl port. Figure 16 shows various ways of placing the intake ports as considered in the early stages of development.

In the A series shown, the intake valves are placed at right angles to the engine, while in the B series, they are placed parallel in a manner similar to normal gasoline engines. To evaluate the performance of each port variation, we employed a continuous flow test using port models, with three-dimensional numerical analysis.

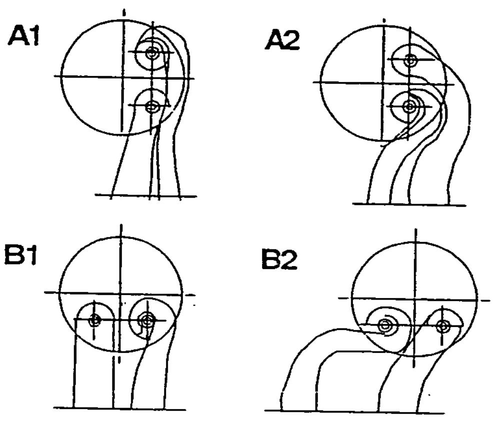


Figure 16. Schematic Diagram of Intake Port Variations

Figure 17 shows experimental results of the continuous flow test using port models. The performance of the helical and tangential ports are taken separately and when they are combined (twin), and are shown as mean swirl ratio and mean flow coefficient.

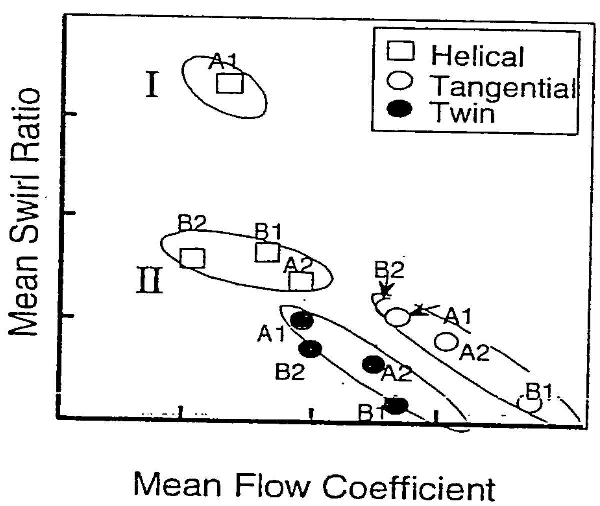


Figure 17. Flow Characteristics of Port Models
( Measurement )

As seen from the figure, intake characteristics for the tangential port and the twin ports are distributed above the straight trade-off line.

In the helical port, on the other hand, the mean flow coefficient is divided into two groups reflecting quite different mean swirl ratios.

With these results the helical port corresponding to group I was therefore selected as the intake port with the task of generating high swirl, and the A1 port variation was adopted to combine this helical port with a tangential port that provides a high flow coefficient.

We also used numerical analysis. In this numerical analysis, we used Star-CD and assumed steady-state conditions imitating swirl evaluation equipment.

Figure 18 shows swirl control characteristics for this port. By opening and closing the swirl control valve (SCV) installed at the input of the tangential port, the swirl ratio can be controlled in the range from 3.5 to 12.

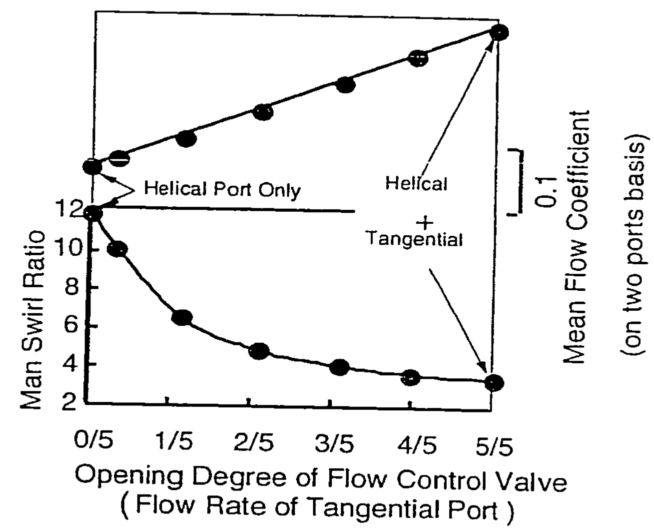


Figure 18. Characteristics of Variable Swirl Controls

## CONCLUSION

Prior to the actual production design, long term research and development such as combustion improvement and port study etc., had taken place. This research made it possible to adopt the new combustion concept to this new DI diesel-engine series. Also lightweight and low-friction know-how from spark ignition engines such as the VQ30DE made these engine series lighter and lower-friction.

Extensive use was made of advanced analytical techniques at every phase of the planning and design process to achieve the target and reduce the development period.

With these efforts, we have succeeded in developing this engine series to meet on time market requirements, such as high power output, low fuel consumption, and low noise level.

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